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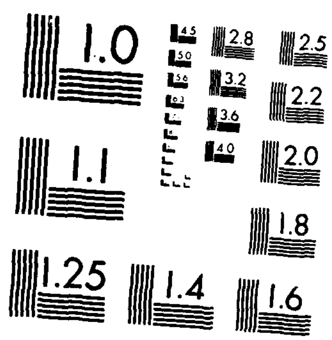
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DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Maryland 20084

A STUDY OF WASTE-HEAT-BOILER SIZE AND PERFORMANCE OF A CONCEPTUAL MARINE COGAS SYSTEM

by

R. K. Muench, D. T. Knauss and J. G. Purnell

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The effect of waste-heat-boiler weight and volume on the performance of an LM2500-based combined gas and steam turbine system (COGAS) is examined. The boiler is a once-through type which is controlled to extract the maximum heat from the gas turbine exhaust and still maintain acceptable minimum wall temperature. At a gas turbine power of 12,000 hp (8.95MW), the boiler without feed-water heating can produce sufficient steam to generate 2840 hp (2120 kW) at a turbine efficiency of 80%. This boiler, including diffuser, weighs

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19,300 ~~lbs~~ (8770 kg) and has a gas side pressure drop of 9 in. H_2O (2.2 kPa). With feed-water heating, the steam power can be increased to 4000 hp (2980 kW) with a boiler weight of 38,3000 ~~lbs.~~ (17,400 kg). The 24 and 33% respective increases in power for these two systems at cruise are translated to 15 and 20% improvement in fuel consumption over the LM2500 gas turbine at the higher power levels of the COGAS system. At the lower power levels, both COGAS systems yield a 20% improvement in fuel consumption over the basic gas turbine.

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ABSTRACT

The effect of waste-heat-boiler weight and volume on the performance of an LM2500-based combined gas and steam turbine system (COGAS) is examined. The boiler is a once-through type which is controlled to extract the maximum heat from the gas turbine exhaust and still maintain acceptable minimum wall temperature. At a gas turbine power of 12,000 hp (8.95MW), the boiler without feed-water heating can produce sufficient steam to generate 2840 hp (2120 kW) at a turbine efficiency of 80%. This boiler, including diffuser, weighs 19,300 lbs (8770 kg) and has a gas side pressure drop of 9 in. H_2O (2.2 kPa). With feed-water heating, the steam power can be increased to 4000 hp (2980 kW) with a boiler weight of 38,300 lbs. (17,400 kg). The 24 and 33% respective increases in power for these two systems at cruise are translated to 15 and 20% improvement in fuel consumption over the LM2500 gas turbine at the higher power levels of the COGAS system. At the lower power levels, both COGAS systems yield a 20% improvement in fuel consumption over the basic gas turbine.

INTRODUCTION

Gas turbines have made simple and compact power plants for combatants a reality. One shortcoming of the gas turbine is its possibly high fuel consumption. The addition of a waste-heat boiler which extracts heat from the gas turbine exhaust gas to operate a bottoming Rankine cycle is one way to improve the fuel consumption. This combined gas and steam turbine system (COGAS) is the easiest way of obtaining improved fuel consumption, requiring only the development of an add-on steam system. The basic performance of the gas turbine is affected only slightly by the additional exhaust backpressure introduced by the waste-heat boiler. Although the steam system itself does not need fuel to produce power, it does add significantly to the volume and weight of the total system. This paper examines the trade-off between waste-heat-boiler size and performance, including the effect of feed-water heating. Feed-water heating is necessary in some instances in order to avoid sulfuric-acid condensation in the boiler. The sulfuric acid in the exhaust gas is produced from the sulfur in the fuel.

The waste-heat boiler utilized in this study is a once-through cross-counterflow type, shown schematically in Figure 1. In this boiler, the feed water is introduced at the gas outlet and is moved through the core until it leaves the other end at the desired condition (superheat). The steam exit conditions are controlled by the feed-water flow rate. A low water flow rate produces high superheat, high minimum wall temperatures and low heat recovery. This control feature is utilized in this study to produce maximum heat recovery at an acceptably lower wall temperature.

The gas turbine utilized in this study is the General Electric Company LM2500, which is nominally rated at 21,500 hp (16.0 MW) for naval applications.

In the study, the size and weight of the waste-heat boiler, with and without feed-water heating, will first be determined. This sizing will be accomplished at an assumed cruise point of the gas turbine. Based upon this boiler sizing, design cases are selected, the performance of which will be determined over their entire power profile.

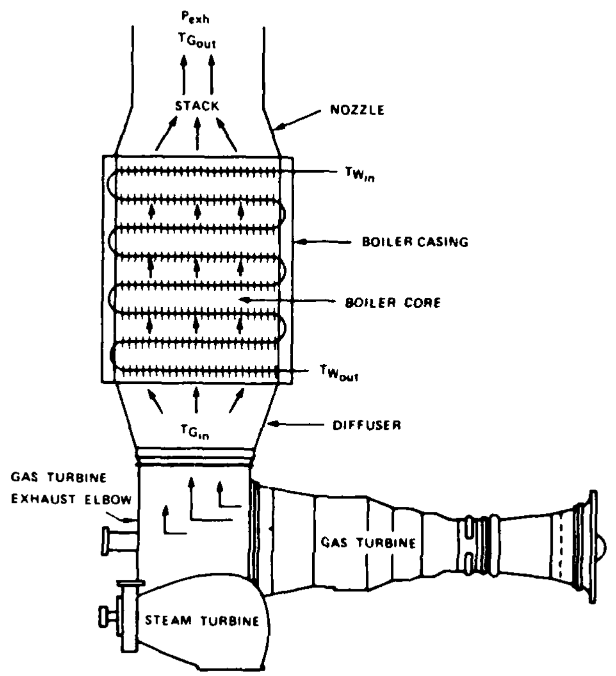

Figure 1 - Once-Through Waste-Heat-Boiler

ANALYSIS

In order to rapidly calculate the size and performance of the waste-heat boiler, a simplified, but relatively accurate, computerized method was developed; it is described in Appendix A. The temperature profile in a once-through boiler is schematically shown in Figure 2. The boiler, as shown in this figure, is divided into four sections (economizer, low and high quality evaporators and superheater). The thermodynamic conditions and heat-transfer rates are calculated as a function of the average temperature in each of these sections. The water-side heat-transfer coefficients range from a maximum of 5000 Btu/hr-^{o}F-ft^{2} (28 kW/(m$^{2}\cdot^{o}$C)) in the low-quality evaporator to a minimum of 200 Btu/hr-^{o}F-ft^{2} (1.1 kW/(m$^{2}\cdot^{o}$C)) in the high-quality evaporator and superheater. The gas-side heat-transfer coefficient is on the order of 20 Btu/hr-^{o}F-ft^{2} (0.1 kW/(m$^{2}\cdot^{o}$C)).

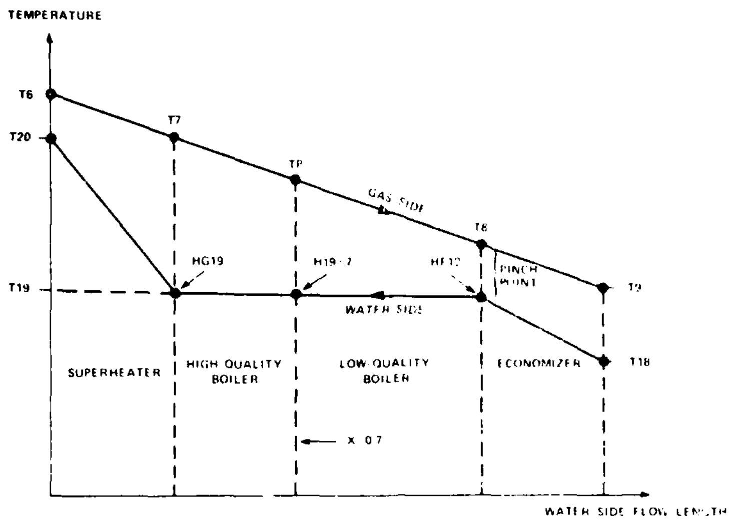

Figure 2 - Boiler State-Point Designations

The calculation starts with the given gas-side temperatures and mass flow (a function of assumed gas-turbine cruise conditions). With assumed steam-side superheat and feed-water temperatures, the total heat transfer and water/steam mass flow are calculated. Also calculated is the heat transfer in each of the boiler sections. This heat transfer is then converted to a heat-transfer area and, ultimately, a tubing length from the known characteristics of the boiler tube bundle. The analysis relies on the theoretical developments and heat-transfer data provided by Kays and London [1]. The boiler-tubing weight is then added to the weights of the boiler casing, water, diffuser, and nozzle to yield a total boiler weight. In this particular model, the core and casing weights each represent approximately 1/3 of the total boiler weight. See Figure 1 for identification of these components.

The gas-side pressure loss through the core is also calculated, based upon the pressure-loss coefficient of the particular boiler-tube geometry. Added to this loss are the diffuser and nozzle pressure losses. Also added is a 2 in. H_2O (0.5 kPa) pressure-loss allotment for the remainder of the exhaust system. The diffuser is assumed to have a rectangular cross-section; it diffuses the exhaust gas from the gas turbine exhaust exit to the inlet of the boiler. The combined length of the diffuser and boiler is taken to be 8 feet (2.4m). If the diffuser included-angle exceeds 70 degrees, the overall height of the system is doubled to 16 feet (4.9m). This is done to make the boiler system compatible with ship deck-heights, which are approximately 8 feet (2.4m).

Wall temperatures at various points in the boiler are also calculated. One of the most important wall temperatures is located at the water inlet of the economizer. This minimum temperature must be maintained above the sulfur-acid dew point. In this study the limit was taken as

275°F (135°C), which is slightly above the dew point of the exhaust gas produced by the LM2500 using fuel with 1% sulfur [2]. The standard Navy marine diesel fuel [3], which is used in gas-turbine-powered combatants, can have a maximum sulfur content of 1%.

The tubes used in this boiler study are nominal 3/4 inch (1.9 cm) diameter with nine fins per inch (3.5 fins per cm) and are arranged in a staggered tube bank (see configuration CF-9.05-3/4 J (a) in reference [1]). The condenser hot-well condition is 115.7°F (46.5°C), corresponding to a condenser pressure of 3 in. Hg (10 kPa). Without feed-water heating, the condenser water is fed directly to the economizer inlet. In other situations, heated-feed-water temperature increments are specified, and the steam flow needed for this feed-water heating is extracted from the turbine at a specified extraction pressure. The steam-turbine power is also corrected for the portion of the extracted flow not expanded through the turbine.

The outputs of interest to the waste-heat-boiler sizing are primarily boiler weight, steam power, boiler gas-side pressure drop, boiler height and pinch point. Additional outputs are needed for the performance evaluation over the entire load line of the COGAS system. These outputs are total heat-transfer area, distribution of heat-transfer area over the various boiler sections, normalized steam turbine flow rate, extraction fraction, boiler gas-side pressure-loss coefficient, boiler frontal area, and steam/gas turbine speed ratio.

The performance of selected design cases over the entire power range is calculated with a state-point matching technique. The model used for this COGAS simulation was originally programmed for the LM2500 gas turbine [4] and later modified for a COGAS system using a recirculating waste-heat boiler [5]. The method was further modified for the once-through boiler used in this application. Improvements in the heat-transfer-calculation procedures were also included. The steam-system-performance portion of the program is described in more detail in Appendix B.

The performance of various components making up the system is calculated as a function of the independent variables or state points. The resulting conditions, such as mass flow and heat transport between components, may not agree, giving rise to errors. The independent variables are then iterated to minimize these errors. The gas-turbine calculations are based mostly on tabulated data of the component performance. The steam-system calculations are based on tabulated heat-transfer coefficients and parameters obtained from the above boiler-sizing method.

The calculation method for the boiler performance is essentially the inverse of the boiler-sizing method discussed above. Instead of sizing a boiler (total heat-transfer area) for a particular performance and certain gas-side input conditions, the program calculates the performance for a given total heat-transfer area and gas-side input conditions. The program also calculates the effect of flow rate on the

performance of the fixed-geometry steam turbine. The effect of turbine speed and inlet pressure on turbine efficiency is included. The steam-turbine speed is set by the gas-turbine speed through a fixed gear ratio. The program can handle both the case without feed-water heating and the case with feed-water heating by steam-turbine extraction. In the latter case, the fraction of steam mass flow extracted and the steam turbine inlet-to-extraction pressure ratio are held constant. Feed-water temperature is then calculated from a heat balance on the feed-water heater.

The outputs of interest are the power generated by the steam turbine and the resultant system specific fuel consumption. The waste-heat boiler conditions, especially the location of the various regions within the boiler, and the steam-turbine inlet conditions (mass flow, temperature and pressure) are also of interest when trying to explain the resulting rematching of the steam system.

RESULTS

Waste-Heat-Boiler Sizing

For the purpose of sizing the waste-heat boiler, it is assumed that the engine cruise conditions are of interest. After all, most of the system's operating time will be spent near this point. It is assumed that the cruise power level of the LM2500 is 12,000 hp (8.95 MW). Additional engine conditions and nominal steam-cycle conditions are given in Table 1.

Table 1. Nominal Cruise Conditions for the Waste-Heat-Boiler Sizing

Engine Conditions		
Power	12,000 hp	(8.95 MW)
Exhaust Mass Flow	100 lb/sec	(45.4 kg/s)
Exhaust Temperature	796°F	(424 °C)
Inlet Loss	4 in. H_2O	(1.0 kPa)
Inlet Temperature	59°F	(15°C)
Exhaust Gas Dew Point	275°F	(135°C)
Nominal Steam-Cycle Conditions		
Saturation Pressure	300 psia	(2.06 MPa)
Superheat Temperature	700°F	(371°C)
Condenser Pressure	3 in. Hg	(10.1 kPa)
Extraction Pressure	50 psia	(344 kPa)
Steam-Turbine Efficiency	0.80	

Figure 3 shows the effect of the boiler gas-side temperature drop on the weight of the boiler without feed-water heating. Elimination of the feed-water heater was thought to be desirable because of the bulk and weight associated with it. The steam system will require an on-line storage volume for the feed water. This might be handled by an oversized condenser hot well.

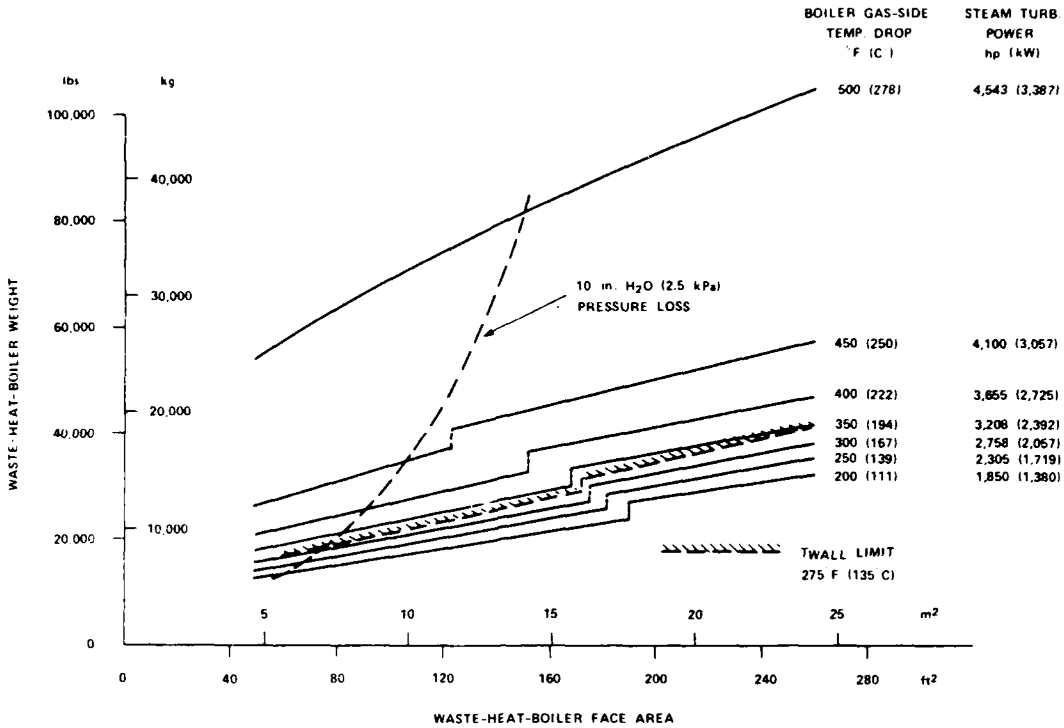

Figure 3 - Effect of Boiler-Exhaust-Gas Temperature on Boiler Size (Weight)

As can be expected, the boiler weight increases with increasing gas-side temperature drop across the boiler. The steam-turbine power (assumed steam-turbine efficiency of 80%) increases linearly with boiler temperature drop. At first, the weight increase is less than the associated increase in steam power, but even as the weight increases become proportionately larger, they never overbalance the steam power from the more effective boilers. Before this happens, the minimum wall temperature (at the gas exit of the economizer section) reaches the sulfuric-acid, dew-point limit. The larger (more effective) boilers will encounter sulfuric-acid condensation in the economizer section. The dewpoint has been assumed to be encountered at 275°F (135°C) (LM2500 gas turbine using 1% sulfur fuel [2]).

Increasing the frontal area of the boiler has a significant effect on the boiler gas-side pressure drop, as shown in Figure 4. This is a result of the inverse relationship between the area and velocity. The boiler weight increases with increasing frontal area (Figure 3) since, with decreasing velocity, the heat-transfer coefficient decreases on both sides. A reasonable exhaust pressure loss may be on the order of 10 in. H_2O (2.5 kPa) at cruise. The total pressure loss consists of the calculated core and diffuser losses and 2 in. H_2O (0.5 kPa) for miscellaneous losses.

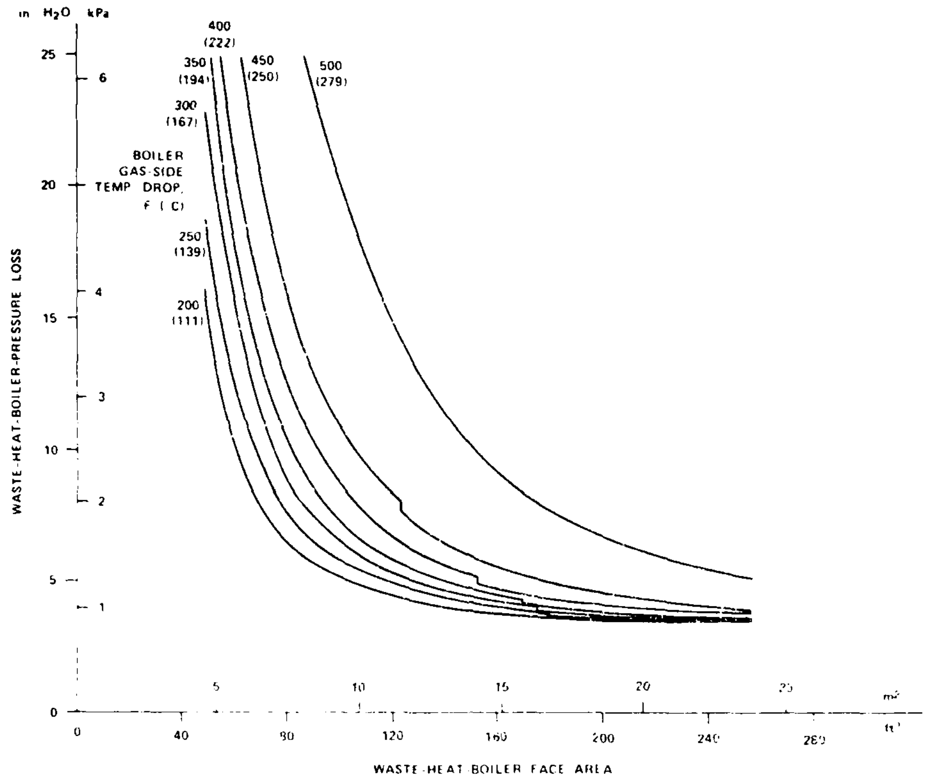

Figure 4 - Effect of Boiler-Exhaust-Gas Temperature on Boiler Size (Pressure Drop)

The 10 in. H_2O (2.5 kPa) pressure-loss line is also shown on Figure 3. To the right (larger frontal areas), the pressure loss is lower. At larger frontal areas, the angle of the diffuser exceeds 70^o and, as stated before, it is then assumed that the overall height of the boiler and diffuser increases from 8 feet to 16 feet. When this happens the weight of the diffuser increases, accounting for the jump in weight seen in Figure 3. The more detrimental effect is the doubling of the boiler volume. Therefore, it is of interest to select boiler configurations to the left of the discontinuous change in weight.

If one stays within these three limitations: (1) pressure loss of less than 10 in. H_2O, (2) boiler height of 8 ft (2.4 m), and (3) wall temperatures of more than 275^oF (135^oC), then the recoverable exhaust heat amounts to 2840 (2120) to 3020 hp (2250 kW), or less than 25% of the assumed LM2500 gas-turbine cruise power level. Figure 3 showed that additional power could be extracted if a larger gas-side temperature drop across the boiler could be taken without exceeding the dew-point temperature limit. The only way to obtain this additional power, in the case of the once-through boiler, is to preheat the feed water.

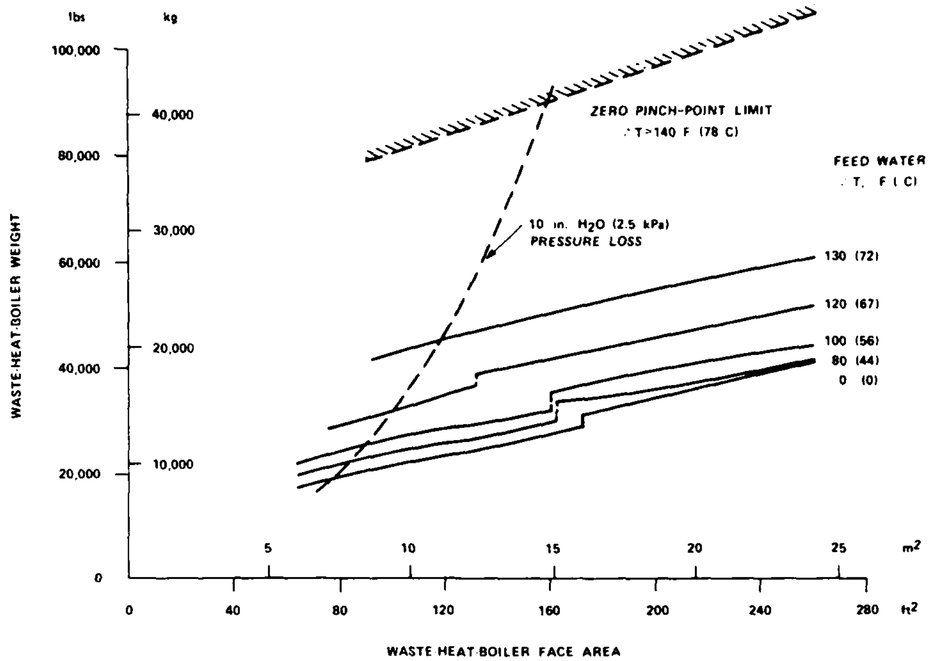

Figure 5 - Effect of Feed-Water Heating on Boiler Size (Weight)

Figure 5 shows the effect of feed-water heating on boiler size as various increments of feed-water heating are provided. Again, the region of interest is between the pressure loss of 10 in. H_2O (2.5 kPa) and the discontinuous change in weight. Initial increments in feed-water heating result in very little gain in steam power. It can be shown that the nominal feed-water temperature of 115.7°F (46.5°C) is near the minimum-heat-transfer level of a given size waste-heat boiler. From Appendix A, the following equation can be written for the total heat transferred as a function of the feed water (TFEED) and wall (TW) temperatures:

$$QTOT/(MG \cdot CP) \cong T6 - TW - (TW - TFEED)(HI/HO)(AI/AO)(1/FE)$$

As the feed-water temperature increases, the driving potential (TW-TFEED) decreases, but the internal heat-transfer coefficient (HI) increases. These effects lead to minimum heat transfer at a feed-water temperature of approximately 130°F (54°C) for a 100 ft^2 (9.3 m^2) frontal area, see Figure 6. This does suggest that better waste-heat-boiler configurations, utilizing different tube geometries, do exist for the case of zero feed-water heating. It also suggests that there are different optimum configurations for the cases with and without feed-water heating.

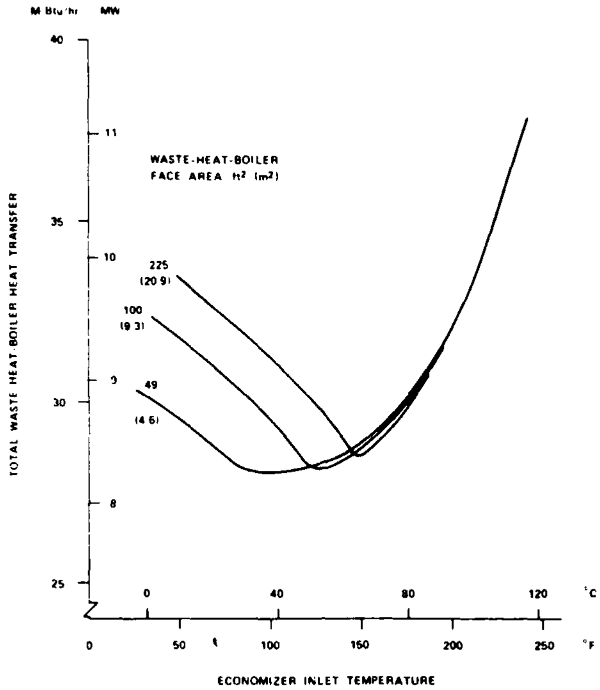

Figure 6 - Effect of Economizer Inlet Temperature on Total Waste-Heat-Boiler Heat Transfer

One of the best ways to summarize the weight, pressure loss, and power trade-off, in a total-system context, is shown in Figure 7. The figure, with exhaust pressure loss and steam power as ordinate and abscissa, respectively, has constant system specific fuel consumption plotted as light straight lines. The waste-heat-boiler data is plotted with heavy lines, the constant feed-water heating conditions are shown as solid lines, and the constant waste-heat-boiler weight as dashed lines. Also, shown as a limit line, is the transition between the 8 (2.4) and 16 feet (4.9m) boiler-system heights. A feed-water increment of 140°F (78°C) corresponds approximately to the zero-pinch-point limit of the waste-heat boiler. The maximum steam power which can be extracted at this point is 4250 hp (3170 kw) or 35% of the assumed cruise rating of the gas turbine.

It is noticed in Figure 7 that the case without feed-water heating shows an increase in steam-turbine power at the lower exhaust-pressure losses, while the feed-water heating cases show the opposite trend. This phenomenon can again be explained with the aid of Figure 6. The zero feed-water heating case, 115.7°F (46.5°C), is to the left of the minimum where the heat transfer to the waste-heat boiler increases with larger frontal area (lower pressure drop). All of the feed-water heating cases shown in Figure 7 lie to the right of the minimum (Figure 6) and, therefore, the heat transfered to the boiler increases with decreasing frontal area or increasing boiler pressure loss.

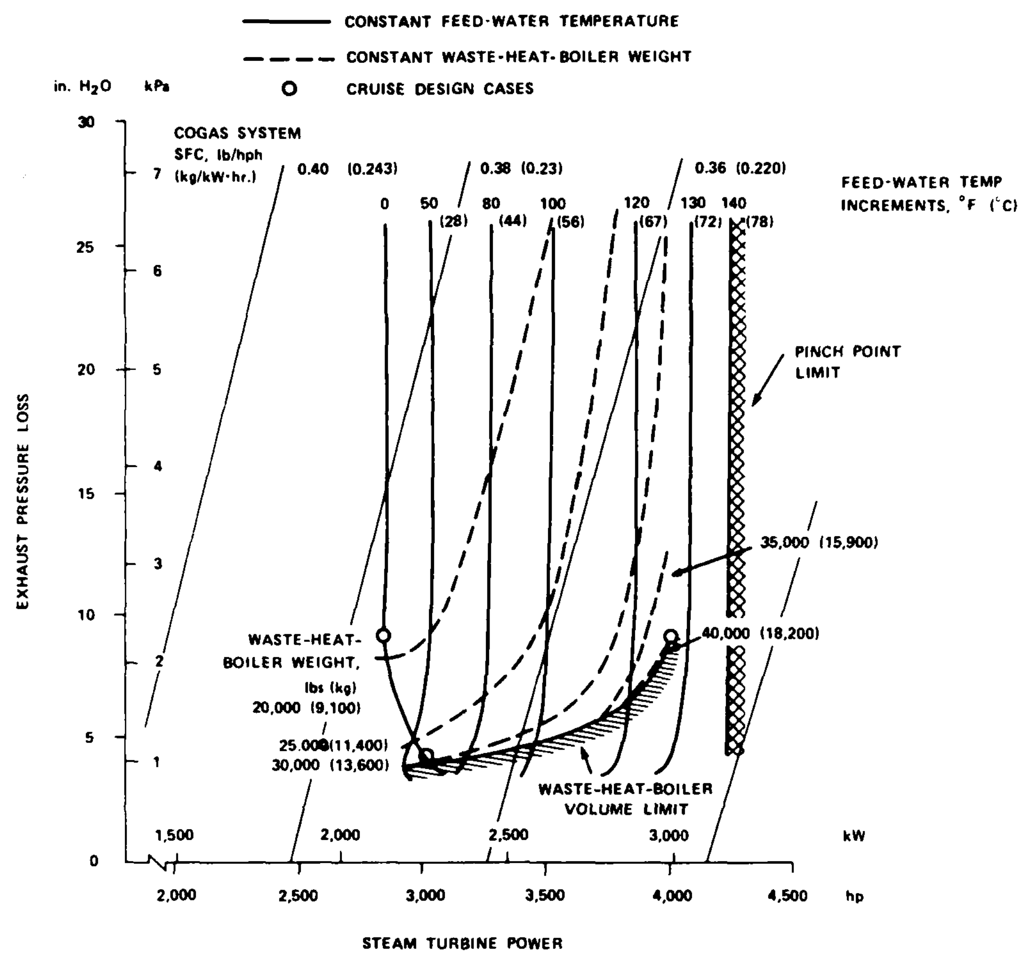

Figure 7 - Waste-Heat-Boiler Sizing Summary

So far, the waste-heat-boiler sizing has been performed at the nominal saturation and superheat conditions, see Table 1. Increasing the saturation pressure shifts the constant feed-water heating lines to higher steam-turbine power, but the weight and limit lines (pinch point and boiler volume) do not change significantly. Higher saturation pressure actually results in a somewhat lower boiler heat transfer, but the Rankine-cycle performance improves with increasing pressure. Increasing the superheat has an effect on the COGAS-system performance that is similar to increasing the saturation pressure. The steam power produced for a given feed-water-heating condition increases, but the weight and limit lines do not shift significantly.

Three different boiler sizes were selected from Figure 7 for further analysis. The first (Case 1) is the lightest and smallest boiler without feed-water heating, which weighs 19,300 lbs (8770 kg) and yields a system specific fuel consumption of 0.376 lb/hp-hr (0.229 kg/(kW.h)) (see Table 2). Even without feed-water heating, the specific fuel consumption can be improved by 1.5% to 0.370 (0.225) by accepting a larger frontal area. This boiler (Case 2) weighs 9,300 lbs (4230 kg) more than the minimum size boiler. More significant improvements can be obtained by accepting feed-water heating. Case 3 is a boiler which results in a 7.5% improvement in system specific fuel consumption over the lightest case. It weighs 19,000 lbs (8640 kg) more than Case 1. It should also be stated that the system weight will go up further due to the feed-water heating equipment and additional piping (which were not sized for this study).

Table 2 - Waste-Heat-Boiler Summary

	Design Cases					
	#1		#2		#3	
Weight, lbs (kg)	19,300	(8770)	28,600	(13,000)	38,300	(17,400)
Dimensions, L*W*H, ft (m)	9.0 * 9.0 * 8.0 (2.7 * 2.7 * 2.4)		13.0 * 13.0 * 8.0 (4.0 * 4.0 * 2.4)		10.6 * 10.6 * 8.0 (3.2 * 3.2 * 2.4)	
Heat Transfer Area, ft^2 (m^2)	13,200	(1230)	20,100	(1870)	36,600	(3400)
<u>Cruise</u>						
Steam Power, hp (kW)	2840	(2120)	3020	(2250)	4000	(2980)
System SFC, lb/hp-hr (kg/(kW.h)	0.376	(0.229)	0.370	(0.225)	0.350	(0.213)
Gas-side Pressure Loss, in. H_2O (kPa)	9.1	(2.3)	4.3	(1.1)	9.5	(2.4)
Steam Flow Rate, lb/hr (kg/h)	22,000	(10,000)	23,400	(10,600)	33,000	(15,000)
Saturation Pressure, psia (MPa)	300	(2.1)	300	(2.1)	300	(2.1)
Superheat Temperature, °F (°C)	700	(370)	700	(370)	700	(370)
<u>Full Power</u>						
Steam Power, hp (kW)	4460	(3320)	4510	(3360)	5870	(4380)
System SFC, lb/hp-hr (kg/(kW.h)	0.327	(0.199)	0.323	(0.196)	0.311	(0.189)
Gas-side Pressure Loss, in. H_2O (kPa)	16.1	(4.0)	6.7	(1.7)	15.2	(3.8)
Steam Flow Rate, lb/hr (kg/h)	38,300	(17,400)	35,500	(16,100)	58,300	(26,400)
Saturation Pressure, psia (MPa)	540	(3.7)	490	(3.4)	550	(3.7)
Superheat Temperature, °F (°C)	730	(390)	830	(440)	710	(380)

COGAS-System Performance

Figure 8 shows the specific fuel consumption of the COGAS system for the three waste-heat-boiler cases discussed above and the baseline LM2500 gas turbine. The specific fuel consumptions of the cruise design points are in agreement with the previous sizing results, Figure 7. At a given power level, the best COGAS case has 20% better fuel consumption than the base gas turbine only. The improvement in fuel consumption decreases slightly with power. The lightest system (without feed-water heating) has a 15% better fuel consumption in the mid-to-high power range than the base gas turbine. At low power, all of the design cases have the same fuel consumption, a 20% improvement over the gas turbine.

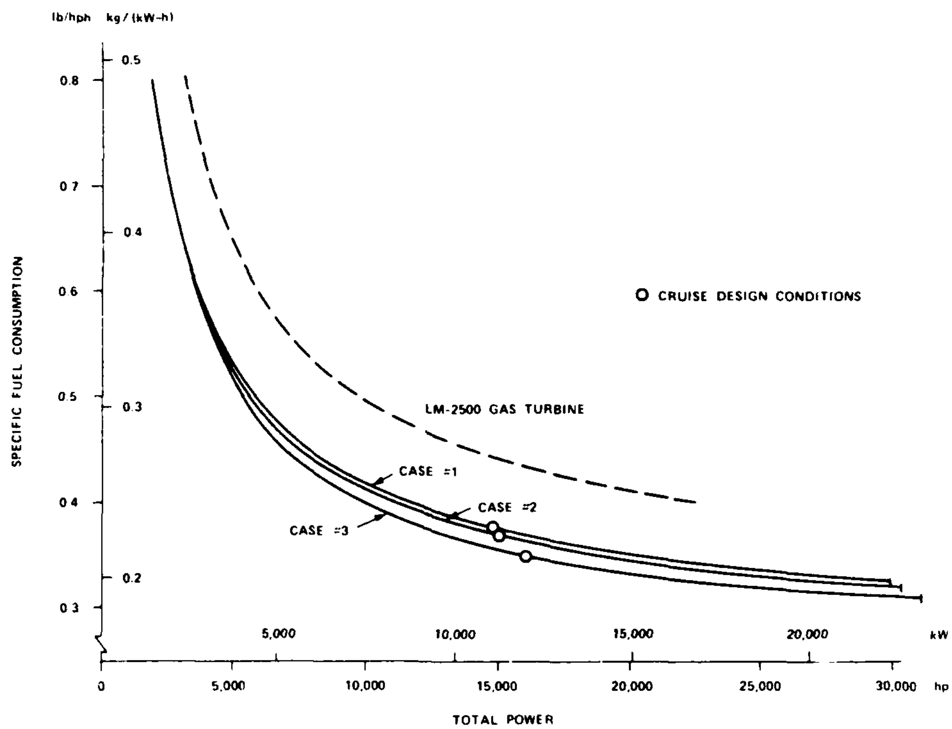

Figure 8 - Specific Fuel Consumption

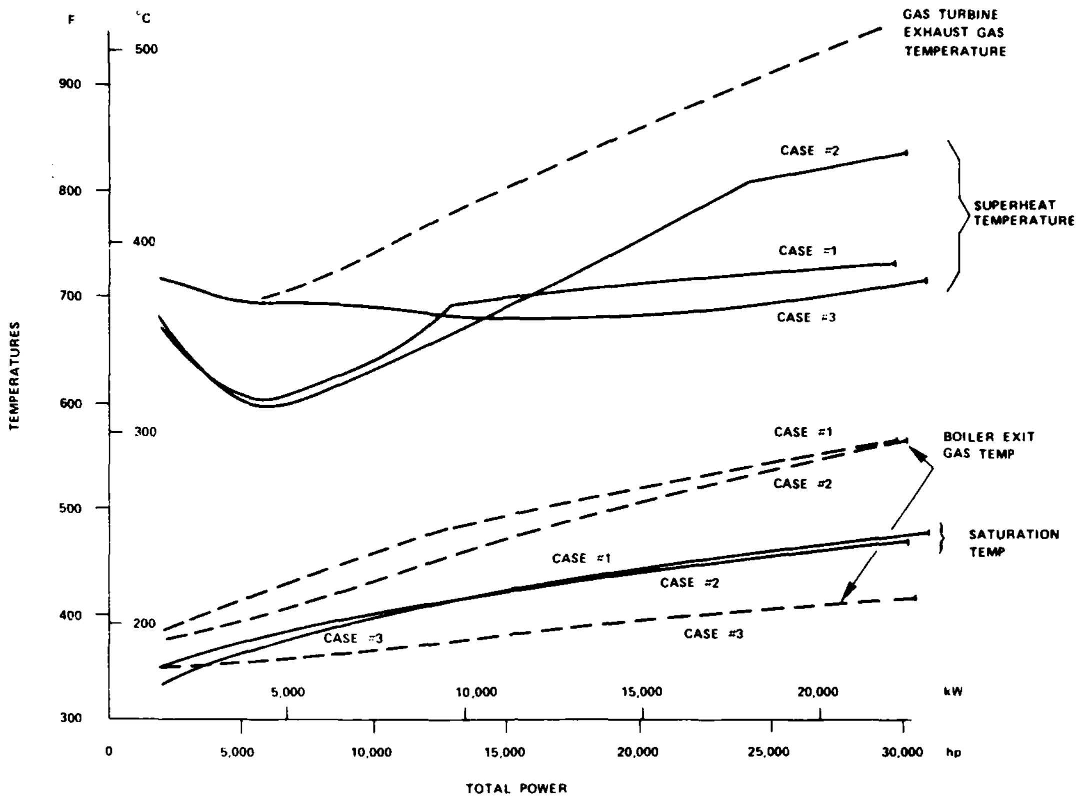

Figure 9 - Waste-Heat-Boiler Temperatures

The performance of the steam system or the improvement of the fuel consumption over the base gas turbine is a function of (1) the amount of heat transferred in the waste-heat boiler and (2) the conversion efficiency of the Rankine cycle. In sizing the waste-heat boiler, attention was focused on maximizing the heat transfer while maintaining the minimum wall temperature. Case 3 allowed more heat to be extracted from the exhaust gas by preheating the feed water; this gave the better performance at the cruise (mid-power) condition. Figure 9 shows the resulting lower boiler-exit-gas temperature of Case 3 in the mid-power range in comparison with the other two cases. At lower power levels, the Case 3 boiler still extracts more heat from the exhaust gas, but its advantage over the other cases is decreasing as power is reduced. This, coupled with the lower pressures at which Case 3 operates, see Figure 10, accounts for its loss of performance advantage at low power, over the other cases. The more rapid decrease in the saturation pressure of Case 3 is a result of the economizer wall temperature limitation. Figure 11 shows the flow rate as a function of power level. Case 3 can pass significantly more water through the economizer in the mid-to-high power range without running into dew-point problems. This is not the case at lower power levels. The mass flow passing through the fixed nozzle area of the steam turbine controls the steam-system pressure level.

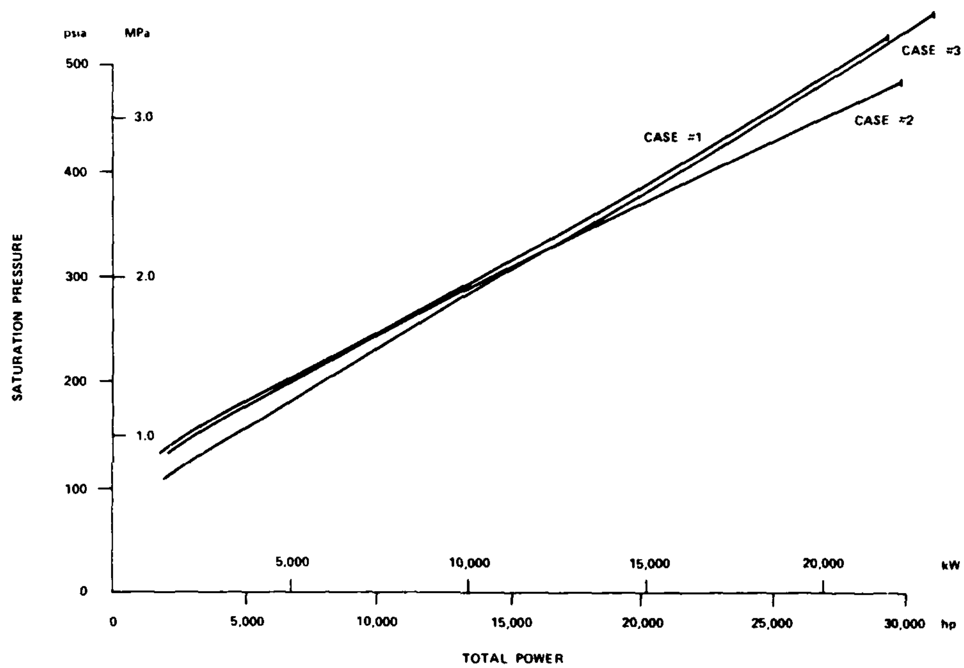

Figure 10 - Waste-Heat-Boiler Saturation Pressures

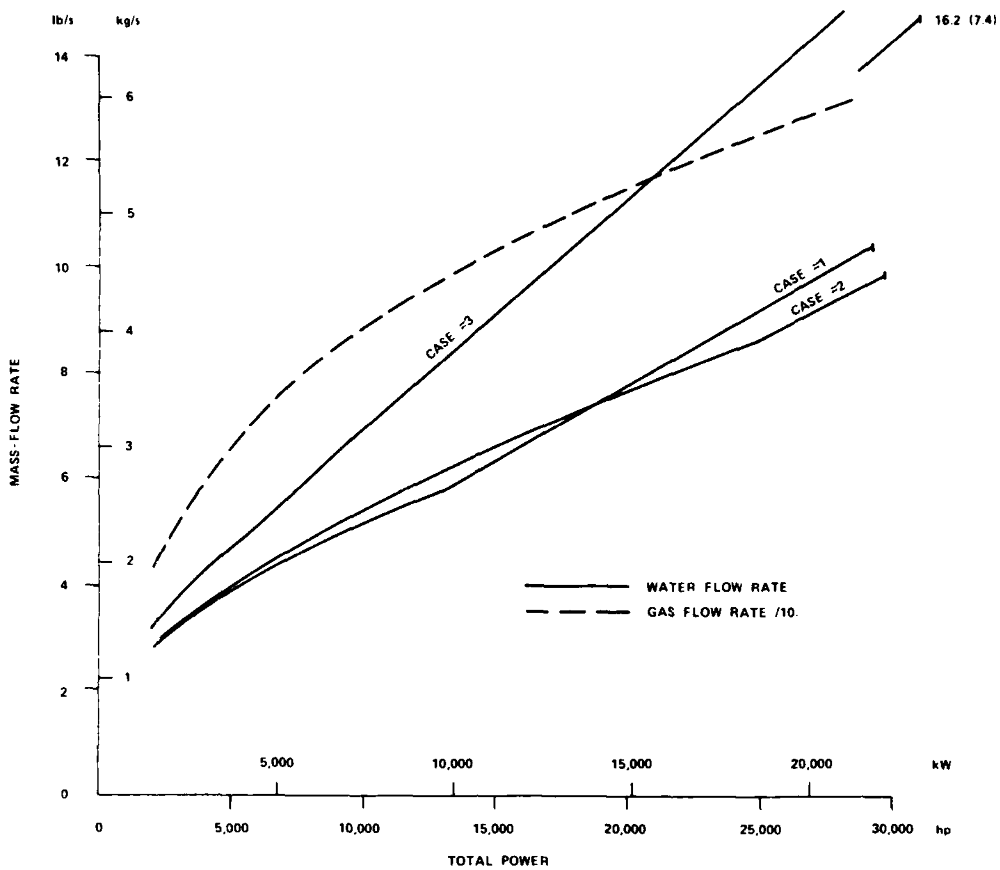

Figure 11 - Waste-Heat-Boiler Mass Flow Rates

Some additional observations about the steam conditions should be noted before concluding this paper. Figure 9 showed some rather severe variations in superheat temperature. The Reynolds number on the waterside in the economizer section falls within the flow-transition region. If a smoother correlation or tabulation were used for the heat-transfer coefficient, a smoother superheat-temperature variation would result. The important item is that in the transition region, the heat-transfer coefficient decreases somewhat faster with decreasing Reynolds number than it does in the turbulent region. Therefore, the proportionately lower heat-transfer coefficients at lower powers allow proportionately larger mass flows without running into the sulfuric-acid-dew point problem. This effect is also seen in Figure 12, which shows how the different boiler regions readjust with power level. Design-case 3 does not operate in the transition region, but it does show a significant increase in the superheater section at low power. The waste-heat boiler, in this case, is too large at the lower power levels. The other two cases would have shown the same behavior except that the economizer sections operate in the transition region at the lower power levels.

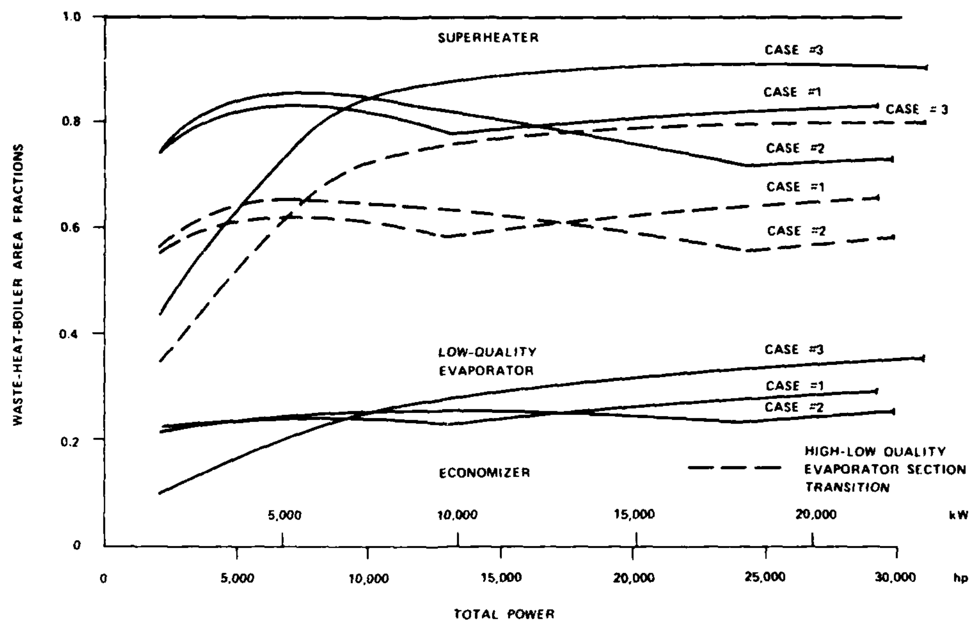

Figure 12 - Waste-Heat-Boiler Area Fractions

SUMMARY

For the particular application chosen, LM2500 gas turbine operating at a cruise rating of 12,000 hp (8.95 MW), the lightest boiler weighs 19,300 lbs (8770 kg), producing sufficient steam to generate 2840 hp (2120 kW) at a turbine efficiency of 80%. This same boiler is capable of producing 4460 hp (3320 kW) at the maximum rating of the gas turbine. In this study the maximum rating of the engine was arbitrarily limited to a power turbine inlet temperature of 1440°F (782°C). The specific weight of this boiler is 4.3 lb/hp (2.6 kg/kW), based on the steam power produced at the maximum rating.

Feed water heating allows more heat to be extracted from the gas turbine exhaust before encountering the dew-point limitation. A boiler with 135°F (75°C) feed-water-temperature increment is able to produce 4000 hp (2980 kW) at the cruise point and 5870 hp (4380 kW) at the maximum rating of the engine. This boiler weighs 38,300 lbs (17400 kg) or 6.5 lb/hp (4.0 kg/kW). It should be remembered that this does not account for the weight of the feed-water heater and additional and larger steam piping. Additional steam power can be obtained, but the boiler weight and volume increased dramatically beyond this point. For the specific cruise rating selected, the boiler can supply sufficient steam to produce 4230 hp (3150 kW) at its zero-pinch-point limit.

The wall temperature at the economizer inlet can be maintained above the acid-dew-point limit by controlling the feed-water flow rate. In the case without feed-water heating, the velocities are sufficiently low that the heat-transfer coefficient is controlled by the transition regime. This allows higher relative flow ratios to be maintained than in the case with feed water heating. The net result is comparable performance at low power for the two cases. It might, therefore, not be advantageous to go to the complexity of feed-water heating, and other methods of maintaining reasonable economizer inlet temperatures while extracting maximum heat from the gas turbine exhaust should be examined.

REFERENCES

1. Kays, W. M. and A. L. London, Compact Heat Exchangers, McGraw-Hill, 1964.

2. Petry, B. R., "Acid Dew-Point Determination for Gas Turbine, Waste Heat Recovery Units," DTNSRDC Report PAS-77-29, Mar 1978.

3. "Fuel Oil, Diesel Marine," MIL SPEC MIL-F-16884F (3 Nov 1965).

4. Bowen, T. L., "Gas Turbine Simulation Technique for Ship Propulsion Dynamic and Control Studies," Fifth Ship Control System Symposium, Annapolis, Md. 1978.

5. Watts, J. W., "Computer Simulation of a Proposed Marine COGAS Propulsion System," SAE Preprint 790648.

6. Jakob, M., Heat Transfer, Vol. I, John Wiley and Sons, New York, 1949.

7. Kern, D. Q., and A. D. Kraus, Extended Surface Heat Transfer, McGraw-Hill Book Co., New York, 1972.

APPENDIX A

WASTE-HEAT-BOILER SIZING

As stated before, the waste-heat boiler is composed of four sections in a cross-counterflow arrangement. Besides the conventional economizer and superheater sections, there is a split evaporator section. This split is necessitated by recognition of the radically different heat-transfer characteristics of the two boiling regions which exist within the evaporator. In the first part of evaporation, which has the lower vapor fraction, nucleate boiling predominates whereas in the high-vapor-fraction region further downstream, the heat transfer is governed by mist flow. The point of division between these two regions is taken to be the location where the vapor quality reaches 70 percent.

The water-side input conditions (feedwater temperature, saturation pressure and steam exit temperature) allow the calculation of the unknown state points on the water side. The gas-side input conditions (mass flow and inlet and outlet temperatures) allow the water mass-flow rate to be determined from an overall heat balance. Once the water flow is known, the amount of heat transferred in each section of the boiler can be determined. By applying the definition of heat-exchanger effectiveness to each section, the area needed to transfer the heat can be established from a trial-and-error solution for the number of heat-transfer units (NTU) and the calculation of the overall heat-transfer coefficient. The sizing of the heat exchanger is completed by converting the area to a corresponding number of passes and total tube length. Other parameters of interest, such as gas-side pressure drop, wall temperatures, inlet diffuser size and pressure loss, and boiler and diffuser weight are also calculated.

The steam tables are used to calculate the various enthalpies, densities, and temperatures needed at the inlet and outlet of each boiler section. By employing the state-point notation given in Figure 2, the enthalpy at the 70-percent-quality interface in the evaporator can be found from

$$\text{H19X7} = \text{HF19} + 0.7\ (\text{HG19} - \text{HF19})$$

where HG19 and HF19 are the enthalpies of the saturated vapor and liquid, respectively. The gas-side energy equation defines the total heat transferred as

$$\text{QTOT} = \text{MG} \cdot \text{CP}\ (\text{T6} - \text{T9})$$

where MG and CP are the mass flow and constant-pressure specific heat of the gas, respectively, with CP evaluated at the average gas temperature. Once the total heat transfer is known, the mass flow on the water-side can be found from

$$\text{QTOT} = \text{MW}\ [\text{CPE}\ (\text{T19} - \text{T18}) + (\text{HG19} - \text{HF19}) + \text{CPS}\ (\text{T20} - \text{T19})]$$

where MW is the water mass flow, and CPE and CPS are the specific heats of the water and steam, respectively, which are evaluated as averages over their corresponding temperature ranges. The above terms for the heat load in the economizer and superheater were specifically chosen over the use of enthalpy differences because of compatibility considerations encountered in a companion program for evaluating the boiler performance in a COGAS power cycle, described in Appendix B.

The above water flow rate can now be used in determining, by heat balances, all of the unknown end temperatures across the various boiler sections shown in Figure 2. As an example, the superheater water-side energy equation is

$$QSHR = MW \cdot CPS\ (T20 - T19)$$

and the gas-side energy equation is

$$QSHR = MG \cdot CP\ (T6 - T7)$$

The above gas-side equation may then be solved for the unknown gas-exit temperature T7 by iterating for the correct average value of CP.

The gas-side heat-transfer area A for each section of the boiler is found by applying the effectiveness-NTU method. The heat-transfer effectiveness (ε) of the section is first found from the general relation

$$\varepsilon = Q/QMAX = Q/[CMIN \cdot \Delta TMAX]$$

where CMIN is the smaller value of the two rate capacities, MG $\cdot$ CP and MW $\cdot$ CPS. Now, it can be shown [1] that, for a given flow arrangement, there is a unique relationship between the NTU, the rate capacity ratio (CR), ε, and the number of passes (PS) made by the water side.

$$NTU = (-PS)\ \ln \{1 + (1/CR)\ \ln [1 - CR(1 - ER^{1/PS})/(CR - ER^{1/PS})]\}$$

where

$$ER = (\varepsilon \cdot CR - 1)/(\varepsilon - 1)$$

Since, initially, PS is unknown, the solution for the NTU is an iterative one. An analogous expression can be derived for the NTU in the boiling sections where CMAX is on the tube side. The resulting expression simplifies since CR = 0, and the NTU is found to be independent of PS. Once the value of the NTU is known, the heat-transfer area A can be found from the definition of NTU which is

$$NTU = U \cdot A/CMIN$$

where U is the overall coefficient of heat transfer, the value of which depends on whether it is multiplied by the inside or outside value of A. Since, in this heat-exchanger problem, the gas-side area is required, the value of U is also based on the gas side.

The evaluation of U can be made by accounting for all of the contributions to the thermal resistance between the two fluid paths. This leads to the relation for the gas-side U, which is

$$1/UO = (AO/AI)(1/HI) + (1/HO)(1/SE) + (ST/KW) + FF$$

where AO/AI is the ratio of the outside and inside heat-transfer areas, respectively, HO and HI are the outside and inside heat-transfer coefficients, respectively, SE is the thermal effectiveness of the outside heat-transfer surface, ST is the boiler tube wall thickness, KW is the thermal conductivity of the wall, and FF is an outside-surface fouling factor. The heat-transfer coefficients are based on the average properties in each boiler section. The outside heat-transfer coefficient is based on the appropriate empirical correlation given in reference 1. This correlation is expressed in terms of the Colburn number for crossflow over various tabulated finned-tube geometries. The inside heat-transfer coefficient used in the economizer, high-quality boiler, and superheater is based on correlations found by Sieder and Tate [6]. This assumes that the heat transfer in the high-quality boiler is controlled only by the vapor phase (as in the superheater). Correlations are available for handling all three flow regimes - laminar, semi-turbulent, and fully turbulent flows. The heat-transfer coefficient used in the low-quality boiler is based on a special boiling correlation, and the evaluation of surface effectiveness, SE, is based on the assumption that the ratio of fin area to total heat-transfer area is sufficiently close to unity to permit replacement of SE with the fin effectiveness. The procedure for calculating fin effectiveness is based on the exact solution for circular fins of rectangular cross section [7].

One other important parameter which is evaluated by the program is the wall temperature at the gas exit of the economizer. In sizing any boiler, only those solutions which yield a wall temperature which is above the acid dew point of the gas are considered acceptable. The solution for the wall temperature is obtained by applying the thermal resistance law to each side of the economizer. By denoting the gas temperature as T9, the water temperature by T18, and the wall temperature by TW, the governing relations for the total resistance on each side can be written

$$RO = 1/(HO \cdot AO \cdot SE) + (FF/AO) = (T9 - TW)/Q$$

$$RI = 1/(HI \cdot AI) + ST/(KW \cdot AO) = (TW - T18)/Q$$

By combining the above equations, Q can be eliminated to give, after some rearrangement,

$$TW = \frac{[ST/KW + AO/(HI \cdot AI)]\ T9 + [1/(HO \cdot SE) + FF]\ T18}{[1/(HO \cdot SE) + FF + ST/KW + (AO/AI)\ (1/HI)]}$$

It should be noted that the denominator is 1/UO and that all properties are evaluated at the fluid temperatures T9 or T18.

Calculation of the steam-turbine power is based on a turbine efficiency of 80 percent. The enthalpy out of the superheater of the boiler (HG20), the isentropic expansion through the turbine to the condenser pressure (PCOND), and the steam flow rate (MW) form the basis for the power calculation. Feed-water heating can also be handled by the program, in which case the feed-water temperature (T18) does not correspond to the condenser pressure (PCOND). Steam is extracted from the turbine at a specified pressure (PEXTR) and used to preheat the feed water. With these two pressures and the specified feed-water temperature (T18), an extraction fraction (EXTRF) of the steam-turbine mass flow is calculated from an energy balance across the feed-water heater. This mass fraction is then applied as a correction to the steam-turbine power calculation.

APPENDIX B

COGAS STATE-POINT MATCHING

A Newton-Raphson convergence technique is utilized to provide the necessary state-point matching. The technique is relatively fast and easily modified. The particular program used here was originally set up for the LM2500 gas turbine [4] and then modified for an LM2500 COGAS system with a recirculating boiler [5]. The program was again modified for the once-through boiler configuration of interest in this study.

The Newton-Raphson convergence technique requires the definition of independent variables (state points), which are usually chosen to include those variables that can't be solved for explicitly. Figure 13 shows the independent variables selected for this particular model. The first eight variables are associated with the gas turbine. These variables and the gas-turbine calculations have been described previously. The next five variables are associated with the steam portion of the system. Associated with the 13 independent variables is an equal number of error equations, which are usually obtained from continuity considerations. The solution is obtained by perturbing the independent variables one at a time to generate a matrix of the change in the error. The matrix is then inverted and applied to the absolute error to calculate new values of the independent variables which will drive all the errors to zero. If the system of equations were linear, only one iteration would be needed. Since the gas turbine and steam system models are highly nonlinear, multiple iterations are needed for convergence.

The 13 error equations used in this model are:

E1 = (P2 - P2A)/P2
E2 = (HPC - HPTI)/HPC
E3 = (MGB - MG3)/MGB
E4 - (MGT1 - MG4)/MGT1
E5 = (T4 - TB)/T4
E6 = (MGT2 - MG5)/MGT2
E7 = (HPL - HPTOT)/HPL
E8 = (P9 - P0)/P0
E9 = (QHBI - QHBO)/QHBI
E10 = (QLBI - QLBO)/QLBI
E11 = (T18-T17)/T17
E12 = (TW - TWLIM)/TWLIM
E13 = (MW20 - MW11)/MW11

The 13 independent variables are defined in Figure 13. The compressor-inlet pressure (P2) is calculated from the ambient pressure (P0) and the inlet pressure loss, which is a function of inlet mass flow. The calculations leading to the next six error equations have been previously described for the LM2500 model [4]; the only difference is that the power in error-equation 7 is the sum of the gas and steam turbine power.

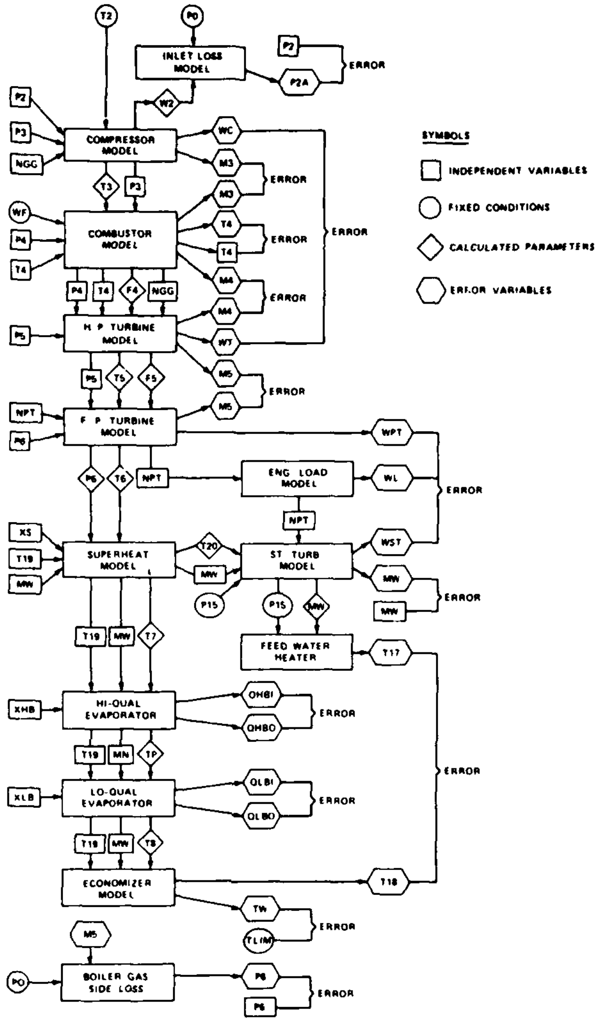

Figure 13 - Flow Chart for COGAS Interation Model

The exhaust backpressure on the gas turbine, P9, is calculated similarly to the inlet loss, with the help of a loss coefficient which was obtained from the waste-heat-boiler sizing routine.

The variables for the last five error equations are a function of the waste-heat-boiler performance, the flow through the steam turbine, and a feed-water-heater heat balance. The method used for this model is essentially the inverse of the boiler-sizing method described in Appendix A. In Appendix A, the boiler heat-transfer area was calculated

as a function of known gas and water-side conditions. In the present case, the water-side conditions are calculated for a known gas flow and temperature at the boiler inlet and a given total waste-heat boiler heat-transfer area. Although the total area of the boiler is constant, the distribution amongst the boiler sections will vary as boiler conditions change. To account for this, the boiler area fractions for the superheater (XSH) and the high (XHB) and low (XLB) quality evaporators were introduced as independent variables. The economizer area fraction (XEC) is also known since the sum of the fractions is unity.

Based on known or assumed temperatures at the ends of each boiler section, the average external and internal heat-transfer coefficients, overall heat transferred, NTU, and effectiveness can be calculated with the same equations developed in Appendix A. In the superheater, the effectiveness is then utilized to calculate the superheat temperature (T20) and the gas temperature T7. Initially, guesses were made for these temperatures to allow transport properties to be calculated. In the high and low-quality evaporator sections, only one effectiveness-temperature relationship is available

$$\varepsilon = (T7 - TP)/(T7 - T19)$$

This still leaves the heat balance in the high-quality-evaporator unsatisfied, and, therefore, it becomes one of the error equations. A similar situation exists in the low-quality evaporator, resulting in another heat-balance error equation. The economizer section is handled similarly to the superheater section, allowing T9 and T18 to be calculated from the two effectiveness equations. T9 does not have to match any other condition. T18 must match the feed-water-heater outlet temperature (T17), resulting in another error equation. The feed-water heater will be discussed below.

The flow through the boiler is controlled by the feed-water pump. A balance between the pump capability and the internal flow resistance of the boiler core could form another error equation. In the present study, it is of interest to maximize the heat recovery without passing below the sulfur-acid dew point. Therefore, an error equation is included which controls the wall temperature at the gas exit of the economizer to the desired value.

The mass-flow continuity must be satisfied at the steam turbine. The steam turbine is a constant-area (no throttling) device. Therefore, the mass flow into the turbine under choked-flow conditions, can be obtained from

$$\text{CHOKCN} = \text{MW11}\sqrt{\text{T11} - 460.}\ /\text{P11} = \text{constant}$$

where CHOKCN is a function of area. The mass flow calculated from this equation is used in the last error equation along with the independent variable MW20.

As mentioned before, the feed-water is preheated in order to extract more heat from the exhaust gas without confronting the sulfur-acid, dew-point problem. As shown in the steam-cycle schematic of Figure 14, the extraction port is located at a given stage of the steam turbine. A portion of the steam is extracted at this point and does not pass through the later stages of the turbine. The extraction port is of constant area, and the flow is choked; therefore, the fraction of the flow extracted (EXTRF) is constant. This fraction was initially determined in the sizing phase of the analysis. The enthalpy of the feed water thus becomes

$$\mathrm{H17} = \mathrm{H15}\ (1 - \mathrm{EXTRF}) + \mathrm{HGEXTR} \cdot \mathrm{EXTRF}$$

where HGEXTR is the enthalpy of the steam at the extraction port. The pressure at the extraction port is related to the inlet pressure by a constant pressure ratio. The steam-turbine power, taking into account the extraction, is

$$\begin{aligned}\mathrm{HPSTRB} = 778/550 \cdot \mathrm{ETA} \cdot \mathrm{MW11}\ [(&\mathrm{H20} - \mathrm{HGEXTRI}) \\ &+ (1 - \mathrm{EXTRF}) \cdot (\mathrm{HGEXTRI} - \mathrm{H13I})]\end{aligned}$$

where H13I is the steam enthalpy which results from the isentropic expansion from T11, P11, to P13.

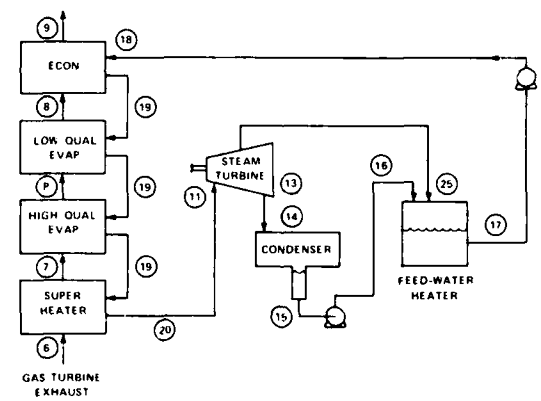

Figure 14 - Steam-Cycle Schematic

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